SURFACE EVAPORATION AND AMMONIA VAPOUR - ABSORPTION REFRIGERATION FOR SOLVING ENERGY PROBLEMS IN COLD STORAGE

by

SANJAY KUMAR

DEPARTMENT OF MECHANICAL ENGINEERING

INDIAN INSTITUTE OF TECHNOLOGY KANPUR

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SURFACE EVAPORATION AND AMMONIA VAPOUR-ABSORPTION REFRIGERATION FOR SOLVING ENERGY PROBLEMS IN COLD STORAGE

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CERTIFICATE

It is certified that the work contained in the thesis entitled **Surface Evaporation and Ammonia Vapour-Absorption Rerfrigeration for Solving Energy Problem in Cold Storage**, by **Sanjay Kumar**, has been carried out under my supervision and that this work has not been submitted elsewhere for a degree.

24/12/96

Dr. Manohar Prasad
Professor
Department of Mechanical Engineering
Indian Institute of Technology Kanpur

December, 1996

Dedicated to -

My Beloved Parents

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Sanjay Kumar
IIT Kanpur
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Synopsis

Name of Student: **Sanjay Kumar** Roll No. **9410525**

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Surface Evaporation and Ammonia Vapour-Absorption Refrigeration for Solving Energy Problem in Cold Storage

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A generalised computer programme has been used to calculate the hourly total cooling load for a cold storage for a given orientation The heat capacity of the wall is also included in terms of time-lag and decrement factor.

Result have been obtained for the case of surface evaporation. It shows that surface evaporation renders about 5 % saving in cooling load. But a saving of about Rs 80,000 per year is found for a normal size cold storage Due to surface evaporation the temperature of roof is almost constant and lesser which reduces thermal stress in building material It has been also found that there is decrease in cooling load due to reduced storage temperature as at low temperature the rate of respiration is minimum.

To solve energy problem in cold storage, the ammonia-absorption system is proposed operating on the primary energy. This reduces operational cost by 40 % and

also helps the cold storage owner to get rid of problem of frequent power break down

A mathematical model is developed for the cooling load prediction. The model is tested for its applicability on a cold storage. The theoretical prediction compares very well with that of the actual cooling load of the cold storage.

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Nomenclature

A_j	area walls and roof, [m^2]
C_{th}	thermal capacity of wall, [kJ]
c	specific heat capacity of building materials, [kJ/kg-K]
c_{pp}	specific heat capacity of potato, [kJ/kg-K]
c_1,c_2	constant
DBT_{out}	dry-bulb temperature, [^oC]
DBT_{in}	wet-bulb temperature, [^oC]
d	declination angle
h	hour angle
h_o	outside convective heat transfer coefficient, [kW/m^2]
h_i	inside convective heat transfer coefficient, [kW/m^2]
IST	Indian Standard Time
I_{df}	diffused sky radiation, [kW/m^2]

I_{dn} direct normal radiation, [kW/m^2]

I_{dr} direct solar radiation, [kW/m^2]

I_r solar radiation reflected from the surrounding surface, [kW/m^2]

I_t total radiation, [kW/m^2]

k_i thermal conductivity of building materials, [kW/m-K]

LCT local civil time

LST local solar time

l latitude

$N_a ch$ numner of air change

Q_b basement load, [kWh]

$\dot{Q}_{infil}$ infilteration load, [kWh]

$\dot{Q}_s$ total solar load, [kWh]

$\dot{Q}_{sen}$ sensible heat load, [kWh]

$\dot{Q}_R$ resperation load, [kWh]

$\dot{Q}_t$ total heat transfer through wall, [kWh]

$\dot{Q}_{total}$ total cooling load, [kWh]

q heat flux, [kW/m^2]

Resp	rate of resperation, [kJ/day-tonne]
T_b	basement temperature, [^oC]
T_{max}	maximum air temperature of a day, [^oC]
T_{min}	minimum air temperature of a day, [^oC]
T_o	outside air temperature, [^oC]
$T_{w,o}$	outside wall temperature, [^oC]
T_i	inside air temperature, [^oC]
T_{sol}	sol-air temperature, [^oC]
TN	tonnage of potato stored
U_j	overall heat transfer coefficient, [kW/m^2]
V_{room}	volume of room, [m^3]
v_i	velocity of air inside room, [m/s]
v_o	velocity of air outside room, [m/s]
$WB_{depression}$	wet-bulb depression, [^oC]
ΔX_i	thickness of wall, [m]
α	wall solar azimuth angle
α_r	absorptivity of roof material
β	solar altitude

γ solar azimuth angle

ϵ emissivity of the surface

η_{sc} saturation efficiency

θ incidence angle

ϕ surface tilt angle from verticle

ϕ_1 time-lag factor

ρ density of building material, [kg/m^3]

λ decriment factor

Chapter 1

Introduction

1.1 Description

India being an agricultural country produces large amount of perishable agricultural commodities of different varieties. Some 20 to 30 % of these products are wasted in the areas where they are grown, because of nonavailability of controlled environment [1].

Of many perishable products the preservation of potatoes is the major application covering 80 % of the cold storages in India [2]. The use of cold storages has been further diversified to store other commodities like fish and dairy products (5 %), vegetables, flowers, spices and other items (15 %).

Cold storages have been proved to be boon to potato cultivators. Their growth in India has been quite rapid being functionally related as

$$F(x) = 3145 + 9256\,4x - 66.646x^2 + 92\,5552x^3 \tag{1.1}$$

where x = Christan era - 1947,as, 1947 has been taken as base in obtaining the above correlation.

With increasing demand of cold storages for various locations, rapid growth in this industry is taking place. In terms of capacity of cold storages, Uttar

Pradesh tops the list with West Bengal at the second place.

Cold storages maintain a low-temperature and suitable humidity to prevent the spoilage of perishable products, make them available in off season and also in other regions where they are not harvested. The growers also get due share of the profit by not selling their seasonal goods at throw away prices.

The rapid developments that are taking place in the cold storage industry has led to greater emphasis on the design aspect. Unlike the arbitrary design of earlier years, new vistas has been opened up in the cold storage designs. These days it is most common practice to construct cold storages and refrigerated warehouses with larger volumes, capable of storing a larger amount of commodities at a lesser cost. Improvements are being made in this field to minimise the functional energy requirements and the cost wherever possible.

Generally, a number of alternatives can be thought of for designing a new system to bring of improvement the existing one. In cold storage the structural cooling load which is dependent on environment is quite substantial apart from commodity load. As we can not reduce commodity load which is fixed one, so our main aim is to reduce structure cooling load.

In this regard, energy saving can be achieved by decreasing the solar radiation falling on walls and roofs of the cold storage by sinking the structure partly or fully below the ground level. Another advantage of sinking the structure partly or wholly under ground is that the underground temperature of the soil is lower than the ambient temperature and moreover the underground temperature fluctuation with time is very small as compared of ambient temperature.

The insulation requirement for underground cold storages is considerably reduced than that for cold storages

above ground level, as a result of lesser heat transmission through underground walls.

The concept of an underground cold storage however has some disadvantages;

like extra digging cost and higher underground constructional cost and unforeseen problems due to seepage of water from the underground.

The shape and orientation of the structure also plays a vital role in energy saving. Since the west and east facing walls are exposed to solar radiation for the longer period. As such the area of these walls should preferably be smaller than those of north and south facing walls. Other orientation can be resorted to if results in decreased structural cooling load and lesser costs The best shape of the cold storage is that which renders minimum surface area for a given volume, rendering reduced cooling load and lesser insulation requirement. The shape of the cold storages is influenced by other practical problems like the constructional cost, maintenance and handling cost etc.

Lastly, energy saving can also be achieved by selecting proper refrigerating machinery and its regular maintenance in order to have a high coefficient of performance. Hence, the optimum cold storage design is envisaged as having both financial feasibility and that of functional energy which renders the overall cost for the cold storage to be minimum.

1.2 Literature Review

The work available in the field of design of refrigeration and air-conditioning systems, about surface evaporation/evaporative cooling and related things pertaining to cold storages and refrigerated warehouses is summarized in the following paragraphs:

1.2.1 Structure Heat Gain

The analysis of cooling load involves parameters such as the ambient air temperature, direct and diffused solar radiations, air velocity, characteristic of enclosing walls and the orientation of the building.

Threlkeld [3] has extensively described the process of heat transfer due to temperature difference between the external and internal environments. The periodic heat transfer through the walls has been explicitly dealt with, using the concept of sol-air temperature His analysis essentially deals with heat transfer through homogeneous walls.

Mackey and Wright [4] have described in detail the periodic heat transfer through composite walls or roofs by reducing a composite wall problem into an equivalent homogeneous wall. The following three methods account for the periodicity of external conditions:

1 Threlkeld's classical approach [3]

2 Transfer function method [5]

3. Finite difference method [6]

Threlkeld's classical approach involves the concept of sol-air temperature and is used in the present work.

In the transfer function method the various components of space heat gain are added together to get an instantaneous total rate of space heat gain. It is then converted into an instantaneous space cooling load through the use of weighting factors called coefficients of 'room transfer function'. The transfer function is nothing more than a set of coefficients that relate to an output function at a given time to the value of one or more driving functions at the time of previous times.

Kadambi and Hutchinson [6] have described an approximate technique to determine one dimensional transient heat transfer through walls and roofs, in the form of the Finite Difference Method. The basic simplicity of approximate method contrasted with analytical techniques is asserted in their work.

A detailed procedure of cooling load calculations, from the point of view of practical designs, is outlined in ASHRAE Handbook of Fundamentals [7]. The

methodology and equations for hour-by-hour load calculations are presented in the handbook.

1.2.2 Economic Design and Optimization

With the limited availability of energy and its consequent increasing cost, it has been necessary to use energy optimally. Extensive work has been done in minimizing the functional energy and cost of cold stores and refrigerated warehouses which are above the ground level or partially underground.

McClure[8] has described a method for the optimum design of building systems to reduce energy requirements.

Bonar [9] has analysed the different parameters and factors that affect the economics of a refrigeration system for a warehouse with reference to the minimization of total costs. Thermal insulation is found to be the foremost factor that affects the functional energy requirement and the energy cost as well.

Hemmings [2] has provided a break-up of the total cost of cold storage as follows:

building work (30%), steelwork and cladding (18%), refrigeration plant and electric bill (21%) and insulation (31%). Thus it can be seen that insulation itself accounts for one-third of the cost. This gives a statistical picture as to the method of controlling the total overall cost of the whole system.

The interested article of Spielvogel [10] demonstrate that the use of more insulation than required can increase the energy consumption and cost.

Gupta [11] has done extensive work on automated optimum design of refrigerated warehouses and airconditioned buildings. He has formulated the design problem as a nonlinear mathematical programming problem and has used multidimensional optimization technique to solve it. His work presents a probabilistic optimum design procedure by considering the randomness of the input

parameters. He has used approximate partial derivative method to evaluate the coolıng and heating load.

Heinge [12] has made a comparative study of the single and multi-storeyed cold storage, and the dependence of insulation thickness on multi-storeyed constructions Single storey cold storages are suitable for places where the cost of land is quite cheap, whereas multi-storeyed cold storage has been found to be more economical provided mechanical handling is used.

The work of Claesson and Holnquist [13] envisages the storage of cold and frozen food in underground mined rock caverns. Their works give an idea of the energy saving from underground storage. They have suggested that such storages can be preferred where disused mines are readily available for underground construction.

Prasad et al [14], in an analytic study, have determined the expression for optimum insulation thickness and sink of an underground cold storage in terms of various non-dimensional parameters. He has determined a quadratic relationship between the approximate optimum sink and insulation thickness.

1.2.3 Evaporative Cooling/Surface Evaporation

The increasing cost of electric energy and frequent power break down have necessiated an economic method of cooling. The evaporative/surface cooling is one of such method.

Leonardo da Vinci built a water powered evaporative cooler for the bed room of his patron's wife for first time. It gave reasonable comfort conditions inside the room.

The first mechanical direct evaporative cooler was developed in about 1932 recently Jain [15] did extensive work in the field of roof surface evaporation. He used gunny bags to keep the roof wet at low cost . He also developed an electronic sensor for automatic starting of the pump for sewetting of gunny

bags. The same is used in the present study for cold storage.

The characteristics of dry tropical climate are given in Jannot Yves [16]. The needs of economic equipment for air cooling are outlined from climatic data and local financial possibilities. For this climatic zone, an acceptable thermal comfort can be maintained all the year round by direct evaporative cooling of outside air. Experimental results obtained by testing a direct evaporative cooler built locally at a compatible cost with local incomes are given.

1.3 Present Study

As these days there is more concern for energy conservation, development of energy efficient system and utilisation of thermal energy directly is the main emphasis in present work to deal with energy efficient system for a long term solution of energy problem in cold storage.

For the accurate evaluation of the cooling load of a cold storage a generalised computer programme has been developed by considering daily outdoor temperature variation based on actual as well as approximate value. The same is compared with actual cooling load.

The surface evaporation on roof of a cold storage renders a saving in the cooling load has been found out to the tune of 5 %. The effect of lower surface temperature has been also studied. At the same time it will reduce the thermal stress in building materials.

The cooling load is also calculated by decreasing the storage temperature. Decreasing the storage temperature , cooling load gets reduced significantly.

Cold storage faces frequent power cut and suffers from frequent breakdowns causing reduced profit and loss of stored commodity. To solve this problem, the replacement of the existing ammonia vapour-compression system by ammonia-water vapour-absorption system is recommended. The energy and economic analysis reveal that by using ammonia-water vapour-absorption system, the

operating cost is reduced by 40 %. The investment for the new system is recovered in 2.5 years when 15 % compound interest is considered.

Economy analysis based on energy cost is carried out for Lithium Bromide-Water Absorption system for upper side and Ammonia Vapour-Compression system for lower side. Unfortunately proposition flouts the very purpose of rendering electric power to bare minimum.

Chapter 2

Evaporative Cooling

Evaporative air cooling occurs in nature near waterfalls, flowing streams over lakes and oceans, under summer showers and even upon wet skin. The classic example of evaporative cooling which we read in elementary school science books is the cooling effect felt, when a moistened hand is waved in the air. The evaporative process simply removes sensible heat (i e., cooling by decreasing the surface temperature) and replaces it with latent heat (i.e., increases the moisture content of aır)[21,22].

2.1 Primitive Evaporative Cooling

Evaporative cooling was probably one of the first mechanical cooling methods used by men. Egyptian painting from 2500 B.C. shows slaves fanning porous clay jars to get chilled water. A wall painting in Herculaneum of 70 A.D. shows a leather water bottle used for cooling water by similar means.Both the American Indians of the South West and the ancient Perscian cooled their tents by damp felt or grass mats. Leonardo Da Vinci built a water powered evaporative cooler for the bed room of his patron's wife.

In the ancient period in India, evaporative cooling was even used to make ice. For this shallow beds dug in the earth were filled with 300 mm. straw, upon

which shallow earthen pans were placed. For still frosty nights, even though air temperature fell no lower than 6°C , ice would formed sometimes 40 mm thick due to adequate refrigeration produced by evaporation and radiation in the night sky. The people here, still use the ancient cocos tatti with greater effectiveness. Doors and windows were replaced by tatties covered with dried khuss-khuss grass. These were kept wet traditionally by coolies, now replaced by pumps with catch basin.

2.2 Modern Evaporative Cooling

In modern days, evaporative coolers are extensively used in textile mills, industrial plants and homes after 1900 A.D. Textile mills were natural applicants as they need both cooling for comfort and high humidity to keep the fibres workable. By 1953, evaporative cooling had become a $ 30 million per year industry, providing people with the means to cope with the summer climate. Before automobile air-conditioning was available, small evaporative coolers were used in the South-West to cool cars · Even today, in India people use khuss-khuss tatties on the roof of the car for evaporative cooling.

The first mechanical direct evaporative cooler was developed in 1932 when house holders assisted their wet-cloth hung windows with electric fans. By 1933, thousands of home made coolers had appeared in Arizona. At first, these were burlap covered wooden frames wet by dripping water and mounted at outside windows. In 1934, burlaps were replaced by excelsior pads.

Today evaporative coolers are commonly used in residential and commercial buildings in tropical region with limited commercial and industrial application else where. There are also application of evaporative cooling in large commercial HVAC systems.

2.3 Theory

Evaporation is described as an adiabatic process, meaning that the total amount of heat in the thermal system remains constant. As water evaporates, the sensible heat content of the system falls, while the latent heat content increases by an equal amount. In other words, dry-bulb temperature falls, but moisture content rises. The limit of temperature reduction is up to the air wet-bulb temperature at the beginning of the process. Surface evaporation can cool up to the wet-bulb temperature. The process stops when the relative humidity approaches 100 %. A simple measure of the potential for evaporative cooling at any given air condition is the wet-bulb depression, or the difference between the dry-bulb and wet-bulb temperature It provides an upper limit of the achievable temperature change by direct evaporation.

The evaporative cooling results from a material phase change, i.e., in particular evaporation of water. The amount of heat absorbed by a material as it changes from liquid to vapour is called the latent heat of vapourisation. For water, the latent heat of vapourisation is given approximately by:

$$(2500 - 2.3 \times \text{ water temperature in } ^{\circ}\text{C }) \text{ kJ/kg} \tag{2.1}$$

The initial temperature of the water being evaporated has some effect on the net heat absorbed. However, for most processes, it can be assumed that each kilogram of water that evaporated, absorbs 2442.5 kJ of heat, corresponding to water temperature of 25 °C.

A perfect evaporative cooling process is illustrated in Figure 2.1 for typical May condition at Kanpur. The outdoor condition (41.5 °C dry-bulb temperature and 22.7 % R.H.) is represented by point A on the psychrometric chart. Air drawn from building interior is at 32 °C and 40 % R.H., represented by C. Outside air is sent through the evaporative system and leaves it at saturation condition, B. This idealized process occurs along a constant enthalpy and stops at 100 % R.H. The humid air at B is then mixed with internal air at C. The final indoor air condition will lie some where along the line from B to C. Its exact location depends on the relative masses of the two air streams. A mixture

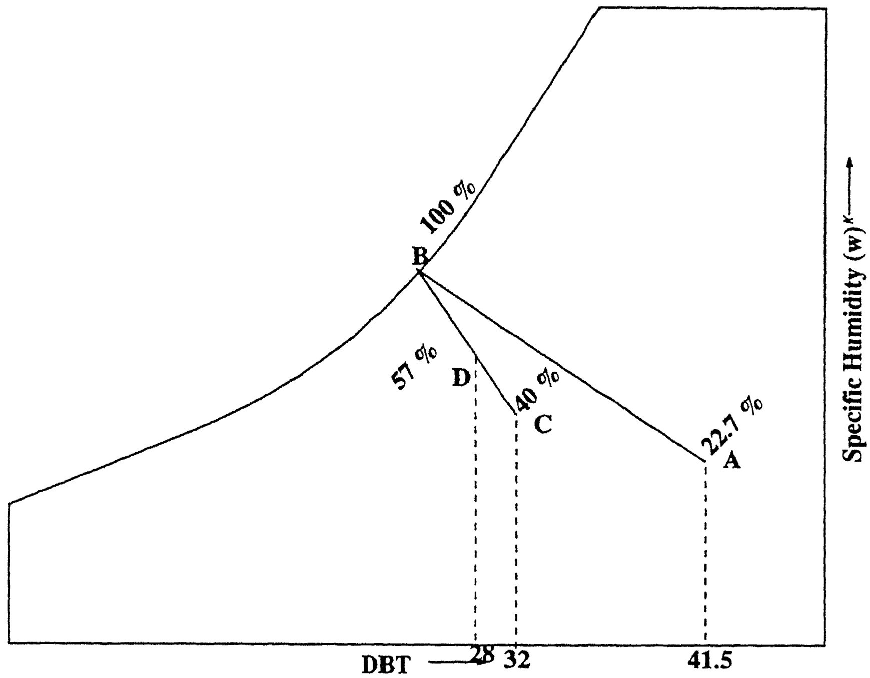

Figure 2.1: Direct evaporative cooling

of one half evaporatively cooled air and one half interior air would provide air at 28 °C and 63 % R.H

2.3.1 Actual Evaporative Cooling Process

Actual evaporative cooling systems have inherent inefficiencies and limitation that prevent them from operating as ideal process. The idealized processes may be considered as the absolute maximum attainable performance. The real system will always provide less cooling i.e., higher temperature.

In an ideal evaporative system, all the air sent through the system is completely saturated with moisture and leaves at 100 % R.H. In actual situation all water is not evaporated and so air is not fully saturated. The capability of an evaporative cooler to cool and humidify the air is measured by saturation efficiency η_{se} as:

$$\eta_{se} = \frac{DBT_{out} - DBT_{in}}{WB_{depression}} \times 100 \tag{2.2}$$

Commercially produced systems provide saturation efficiencies of about 80 %, wıth some time as low as 50 % and other as high as 90 %.

2.4 Evaporative Cooling equipment

Evaporative cooling equıpment may use either a direct or indirect cooling process. In the direct process the water is evaporated directly into the air stream that flows into the conditioned space. In an indirect cooler the evaporation process is separated from the air to be delivered to the conditioned space. Evaporative coolers may also be either single -stage or multi-stage. Multi-stage evaporative systems use two evaporation processes, with the first supplyıng precooled air to a second stage.

2.4.1 Direct Evaporative Cooling

In direct evaporative cooler water is supplied through a float valve to a small reservoir and then flows down through fibrous pads. A fan draws large volumes of outdoor air through pads, where it is cooled by evaporation, and then supplied to the building. This cool and more humid air absorbs sensible heat from the building.

The temperature of air delivered by an evaporative cooler may be estimated by the following equation.

$$T_{supply} = TDB_{out} - (WB_{depression} \times \eta_{se}) \tag{2.3}$$

or,

$$T_{supply} = TDB_{out} - (TDB_{out} - TWB_{out}) \times \eta_{se} \tag{2.4}$$

Using the typical saturation efficiency, i.e., 80 % this equation can be rewritten as

$$T_{supply} = 0.2TDB_{out} - 0.8TWB_{out} \tag{2.5}$$

This equation is useful for judging the applicability of direct evaporative coolers under different conditions. When conditions permit their use, evaporative coolers are much more economical method of cooling than conventional vapor-compression air-conditioners. The power consumption of fan, which ranges from 250 W to 750 W in most residential units. A comparable air-conditioner might consume six to eight times as much electric power. Water consumption in an evaporative cooler is quite significant. A typical evaporative cooler with a saturation efficiency of 80 % requires about 9 liters of water per hour of air flow for each 6 °C of wet-bulb depression.

An evaporatively cooled building operates as an open system where dry outdoor air is continuously drawn into cooler, conditioned, and supplied to the building, providing a complete air change every 2 to 4 minutes.

Direct evaporative coolers humidify the air supplied to a building, rendering the relative humidity of indoors to be always higher than that of outdoors. The successful application of direct evaporative coolers depends on the existence of outdoor humidity levels well below human comfort conditions. But even in arid zones, they are considered as second-class cooling system, because they cannot humidify consistently as done by conventional air-conditioner. Consequently, evaporative systems should be installed with conventional air conditioning systems An airconditioning system which incorporates evaporating cooling and mechanical airconditioning system is called hybrid air-conditioner

2.4.2 Indirect Evaporative Cooling

Indirect evaporating coolers attempt to make use of evaporative cooling process without increasing the amount of moisture in the supplied air to the building. Indirect coolers use a heat exchanger to separate the direct evaporative process from the air to be delivered to the building. A direct evaporative process cools air that flows across one side of the heat exchanger, removing heat, and is then exhausted to the outdoors. The air to be supplied to the building flows across the other side of the heat exchanger and is cooled without receiving moisture

directly.

The cooled water from the cooling tower is circulated through the heat exchanger by a small pump. A fan draws the air through the heat exchanger and supplies it to the building. The system is quite effective in dry climates, often reducing indoor temperature by 10 to 20 °C on a hot and dry climate. A simple indirect cooling process is shown on the psychrometric chart in Figure 2.2. Outdoor air at 41.5°C dry-bulb temperature and 22.7 % R.H. enters the cooler at point A and cooled without the addition or removal of moisture along a horizontal line to point B, 25°C dry-bulb temperature and 57 % R.H.(the actual output conditions depend on the saturation efficiency of the direct evaporative process and the effectiveness of the heat exchanger).

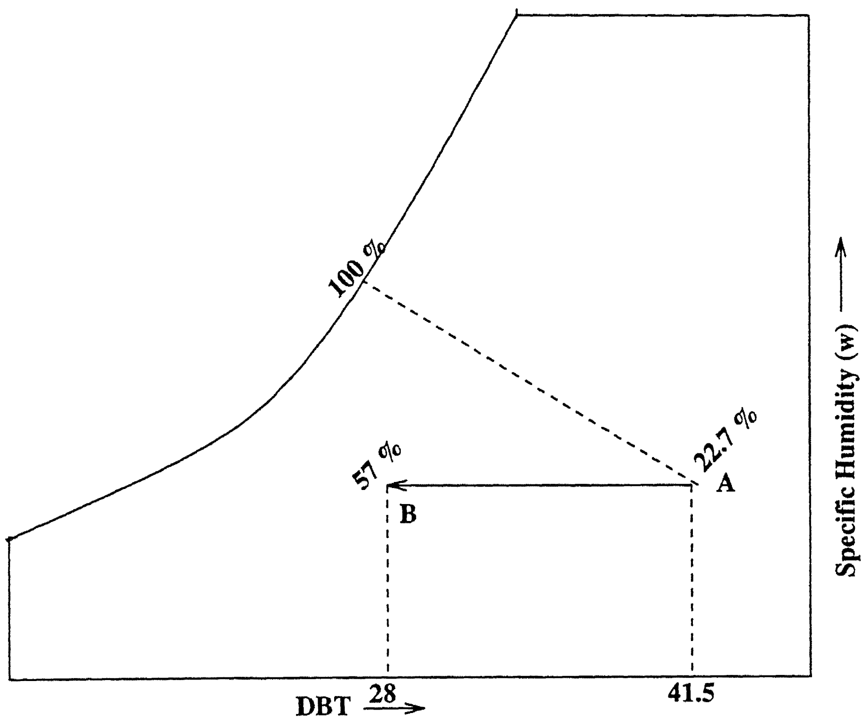

Figure 2.2: Indirect evaporative cooling.

Two-stage evaporative cooling system offers significant improvements in performance over indirect evaporative cooling systems. In a two-stage evaporative system, the cooled air from an indirect cooler is passed through a direct evap-

orative cooler for additional cooling.

2.5 Roof Surface Evaporation

Roof surface evaporation [15], a simple and passive process of cooling buildings is effective, economically viable and practicable technology to solve all the global problems of cooling of buildings. This process not only eliminate major flow of heat to the building through its roof, but it also extracts heat from the building as shown in fig 2 3. Evidently, the surface evaporation brings the roof temperature close to the wet-bulb temperature. Hence it becomes a heat sink and so heat gain by the building through the walls, windows etc. is transferred to the roof, rendering reasonable comfort temperature inside the building. This type of cooling is quite suitable for workshops, industrial buildings, theaters etc

Figure 2.4 shows a schematic diagram of the roof surface evaporation. It comprises a gunny bag layer over the roof, water sprinkler having a pipe connections, a pump to maintain adequate pressure in the sprinkler and a water tank. The evaporative cooling needs small quantity of water upto 9 liters per square meter of roof surface area per day under severe hot-dry weather conditions. Water has to be sprayed uniformly over a thin absorbing material lining providing wick action on the entire roof surface. It must absorb sufficient water and evaporate it quickly. The heat and water absorbing material lining is required to be maintained moist day and night during summer periods for continuous and quick evaporation.

The past manual spraying on empty cement bags for low cost housing schemes has been replaced by highly refined system to meet the requirement of sophisticated modern buildings. This has led to development of automatic spraying device and coirmating of the surface as heat absorptive material lining Such arrangement is reliable and renders better performance. It is the most economic cooling system. Since evaporation takes place at all temperatures and based on well known relationship of latent heat of water versus its tempera-

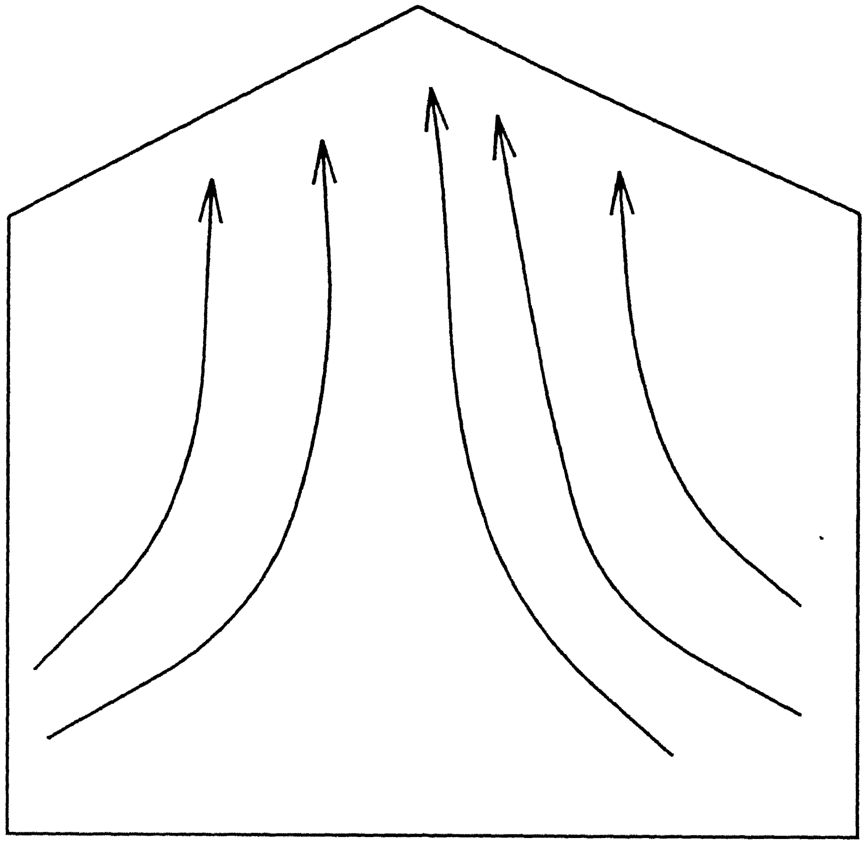

Figure 2.3: Flow of heat towards roof.

ture, it has been found after calculation that 3800 liters of water can produce nearly 900 tons of refrigeration effect if it is properly used in this process as per its basic requirements. Quality of water do not make much difference for this cooling process except replacement of metallic sensing probes in automatic spraying system.

Recent development of hand-operated device and its successful implementation at various places, has made possible for easy adoption of this process by masses even in remote areas where both electricity and tap water may not be available.

The precautions, which have to be taken care of, particularly for water-proofing aspects at the design stage by changing the topography of the roofs more-sloppy pitched or cylindrical dome to avoid stagnation of water on roofs.

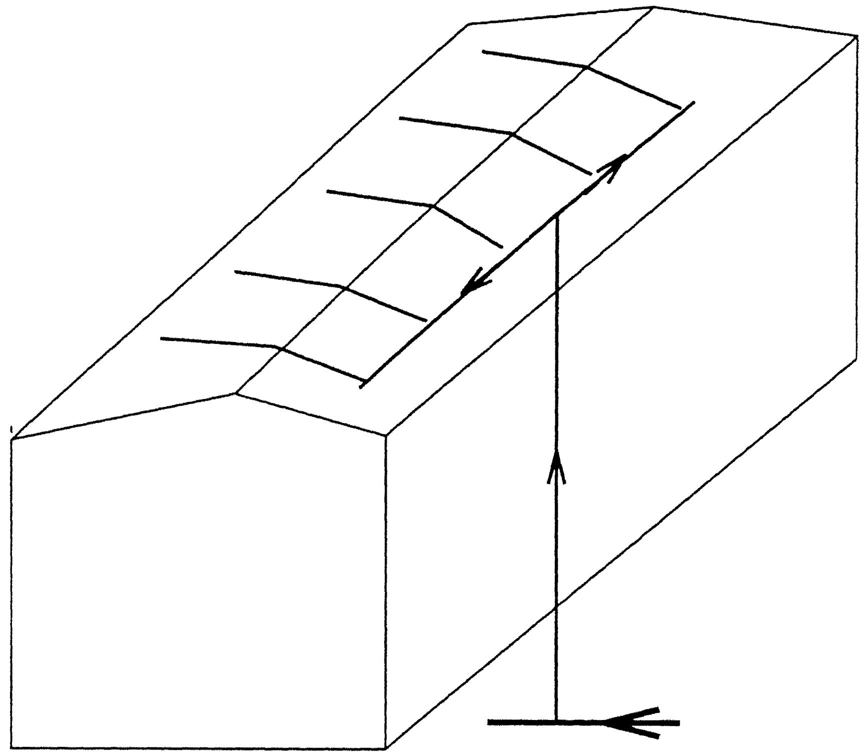

Figure 2.4: A typical arrangement of surface evaporation.

The use of conventional heat insulating materials ,e.g., lime concrete, mud phaska, thermocole having so many practical problem can be straight way eliminated once this process is made integral part of buildings.

In the present work Roof Surface Evaporation is proposed for cold storage.

Chapter 3

Estimation of Cooling Load

3.1 Different Heat Sources :

The heat load in cold storages arises from the following sources:

1 Energy transfer from the surroundings to the system These are:

   (a) Heat transmission through barriers such as walls, doors, ceiling, floor, etc. being caused by the temperature difference existing on the two sides of the barrier.

   (b) The solar heat gain:
   Absorbed by walls or roofs exposed to radiation from the sun and transferred to the inside space.

   (c) Heat and moisture introduced to the conditioned space through infiltration and ventilation air.

2. Heat generation within the confined space. They are:

   (a) Heat load from the product stored inside – sensible and latent

   (b) Heat load from fans, lights and other electrical appliances in use.

3.2 Need for Hourly Cooling Load Calculations

It is well known fact that the ambient air temperature varies during the day and its variation also depends upon the day of a month. Accordingly, the heat load varies over a period of 24 hours This is mainly due to the variation in the solar intensity falling on the earth surface. The load calculation is further complicated by the fact that a wall has thermal capacity, due to which a certain amount of heat passing through it is stored and is transmitted to the inside at some time latter. Therefore, calculation based on instantaneous heat transmission through structure without considering the thermal capacity of the wall, is erroneous. Moreover, the traditional methods of evaluating the cooling load involves various assumptions like the load from each component is constant in a day and maximum value of the cooling load is equal to the individual maximum, which is not correct. Since the occurrence of maximum load at particular time is not the occurrences of the other components. It is proper to adopt an hourly cooling load calculation to facilitate an economic selection of a refrigeration system.

3.3 Heat Transfer Through Structure:

3.3.1 Building Survey

The building survey will include the flowing particulars:

1 Location of the building ,i.e., longitude and latitude

2. Orientation

3 Dimension of the building structure, such as length, breadth , height and thickness of each layer of the building material.

4. Composition of building material and their physical properties.

3.3.2 Hourly Outside Temperature Variation

Figure 3.1 shows the hourly variation in outside temperature for Kanpur. The metrological data reveals that the minimum temperature occurs just before sunshine (may be one or two hours) while the maximum temperature occurs three to four hours after the solar noon [3]. Since, the temperature time relation is available for only a few places, an hourly variation in temperature is predicted based on maximum and minimum temperatures. It has been taken in the following form:

$$T = L + M\cos(15t - N) \qquad (3\ 1)$$

where, T=temperature at any time in °C.

L, M and N are constants, calculated by applying the boundary conditions·

$$\frac{dT}{dt} = 0, \text{for} T = T_{max}\ or\ T_{min}$$

It is assumed that T_{min} occurs one hour before the sunshine and T_{max} occurs 12 hours thereafter.

The comparison of the temperature variation by equation 3.1 with actual variation is also shown in figure 3.1. The data has been taken from the average temperature of three years i.e. 1993, 1994 and 1995 [18]

3.3.3 Solar Radiation

Solar radiation [7] forms the significant part of cooling load in the cold storages The total radiation (I_t), reaching on terrestrial surface is the sum of the direct solar radiation (I_{dr}), the diffused sky radiation (I_{df}) and the solar radiation reflected from the surrounding surface (I_r). The intensity of the direct component is the product of the direct normal radiation (I_{dn}) and the cosine of the incidence angle (θ) between the incoming solar rays and a line normal to the surface as shown in Figure 3.2. Thus,

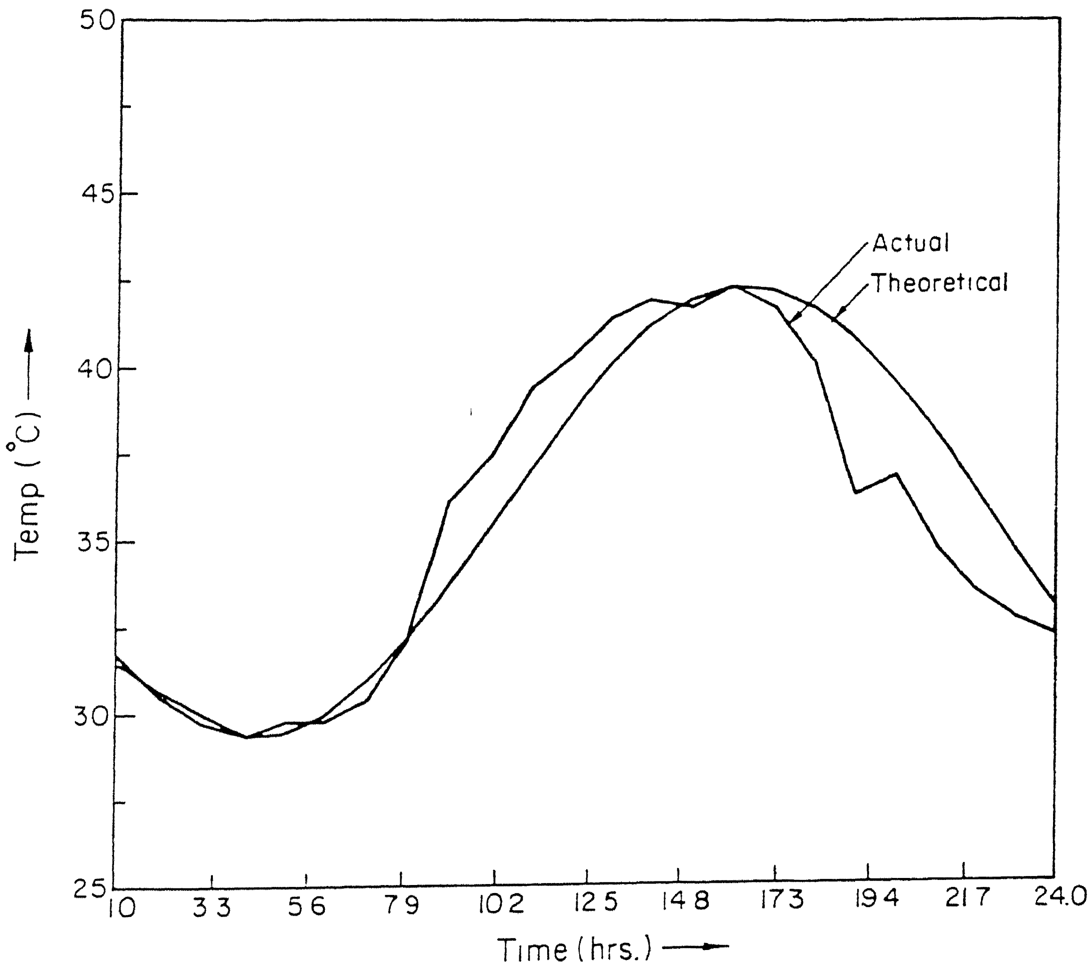

Figure 3.1: Hourly variation of actual and theoretical temperature.

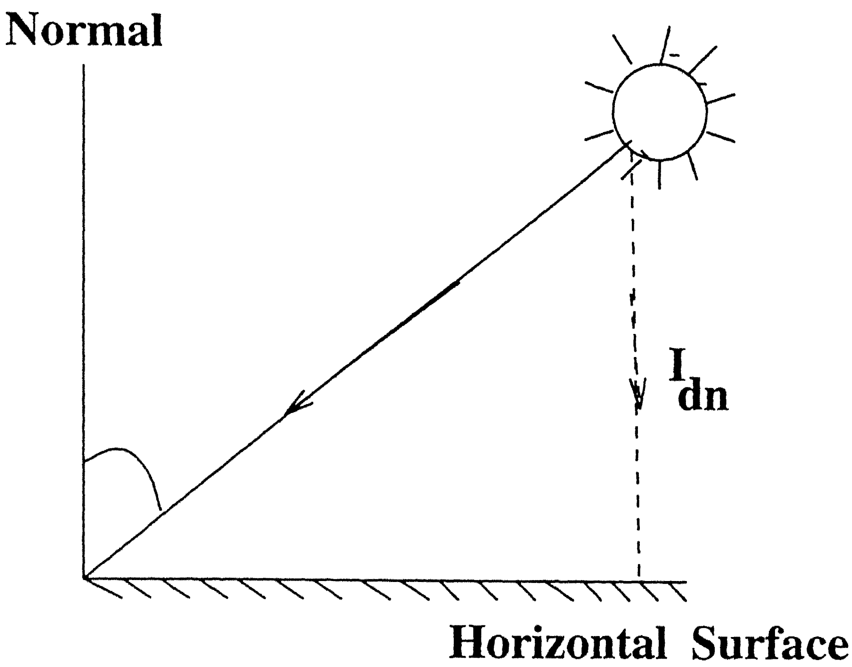

Figure 3.2: Direct solar radiation on horizontal surface.

$$I_t = I_{dn}\cos\theta + I_{df} + I_r \tag{3.2}$$

In the present analysis the reflected radiation (I_r) is neglected.

I_{dn} and I_{df} are given as:

$$I_{dn} = \frac{A}{exp(B/sin\beta)} \tag{3.3}$$

$$I_{df} = C.I_{dn}.F_{ss} \tag{3.4}$$

value of F_{ss}=0.5 for vertical surfaces, 1.0 for horizontal surfaces and $(1+\cos\phi)/2$ for any other inclined surfaces.

The value of A, B and C are tabulated in [7] and are given in Appendix 'B'.

3.3.4 Determination of Incident Angles

The sun's position in sky is most conveniently expressed in terms of solar altitude β, above the horizontal and solar azimuth angle γ, measured from

south. These angle β and γ depends on the latitude of the place l, solar declination d, and hour angle h. β and γ is related with l, d and h by the following expression:

$$sin\beta = cosl.cosd.cosh + sinl.sind \tag{3.5}$$

and,

$$cos\gamma = \frac{sin\beta.sinl - sind}{cos\beta.cosl} \tag{3.6}$$

The hour angle is calculated as follows:

$$LST = LCT + (\text{Equation of time}) \tag{3.7}$$

Where, LST is local solar time and LCT is local civil time which are based on the standard meridian of the country. And due to the irregularities of the earth's rotation, obliquity of earth's orbit and other factors, a solar day is not exactly equal to 24 hours. To counter that effect equation of time is added in LCT, which is given by

$$LCT = [IST - (82.5 - longitude)]/15 \tag{3.8}$$

Where IST is Indian Standard Time which is based on 82.5 meridian (standard longitude for India) passing through Naini near Allahabad

The hour angle is zero at solar noon and increases by 15^o every hour. Hence hour angle,

$$h = (12 - LST) \times 15, \text{ before solar noon} \tag{3.9}$$

$$h = (12 + LST) \times 15, \text{ after solar noon} \tag{3.10}$$

If a surface is tilted by an angle ϕ to the vertical position then the incidence angle θ is given by

$$cos\theta = cos\beta.cos\alpha.cos\phi + sin\beta.sin\phi \tag{3.11}$$

If the surface is vertical, ϕ=0, then

$$cos\theta = cos\beta.cos\alpha \tag{3.12}$$

If the surface is horizontal, $\phi=\pi/2$, then

$$cos\theta = sin\beta \tag{3.13}$$

After knowing the value of h, the values of β and γ are calculated The wall solar azimuth angle, α [7] for various walls are·

$$\alpha_E = \begin{pmatrix} |\pi/2-\gamma| & , & \text{before noon} \\ |\pi/2+\gamma| & , & \text{after noon} \end{pmatrix} \text{east facing wall} \tag{3.14}$$

$$\alpha_W = \begin{pmatrix} |\pi/2-\gamma| & , & \text{before noon} \\ |\pi/2+\gamma| & , & \text{after noon} \end{pmatrix} \text{west facing wall} \tag{3.15}$$

$$\alpha_N = \begin{pmatrix} |\pi-\gamma| & , & \text{before noon} \\ |\pi+\gamma| & , & \text{after noon} \end{pmatrix} \text{north facing wall} \tag{3.16}$$

$$\alpha_S = \gamma,\ \text{south facing wall} \tag{3.17}$$

3.3.5 Sol-air Temperature

The calculation of space cooling load as a result of heat gain through exterior roof and walls involves the concept of sol-air temperature [3]. A heat balance at a sunlit surface gives the heat flux to the surface as:

$$q = \alpha_r I_t + h_o(T_o - T_{w,o}) - \epsilon\Delta R \tag{3 18}$$

ϵ=1, for black body

ΔR=0.063 kW/m^2, for horizontal surface and 0, for vertical surface [7]

The above equation can also be written as

$$q = h_o(T_{sol} - T_{w,o}) \tag{3.19}$$

where,

$T_{sol} = T_o + \alpha_r\ I_t/h_o$ - $\epsilon\ \Delta R/h_o$

3.3.6 Heat Transfer Through Walls and Roof Using Decrement Factor and Time Lag

As building materials have finite thermal capacity, the instantaneous heat transfer through walls and roofs by neglecting thermal capacity of the building materials is not correct The structural detail of walls and ceiling is given in figure 3 3 and 3.4 respectively.

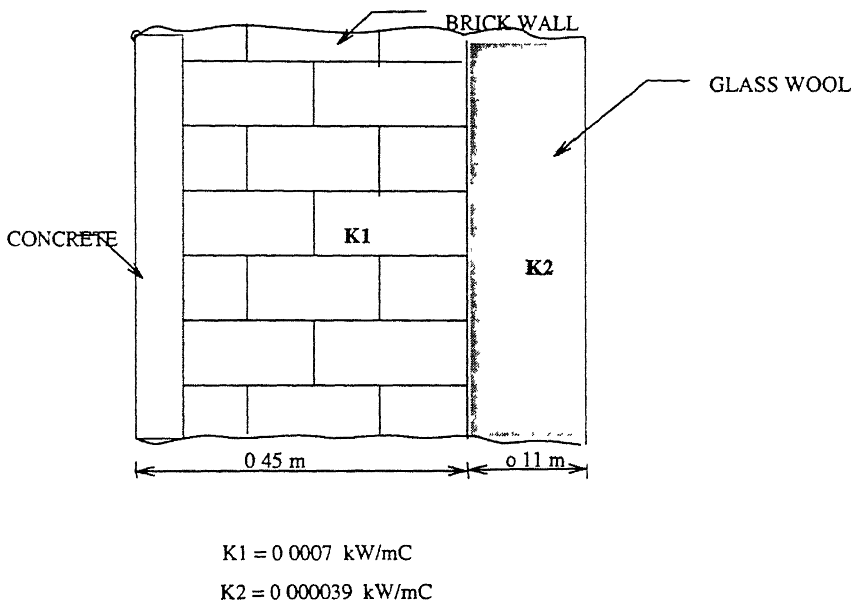

Figure 3.3: Structural detail of wall

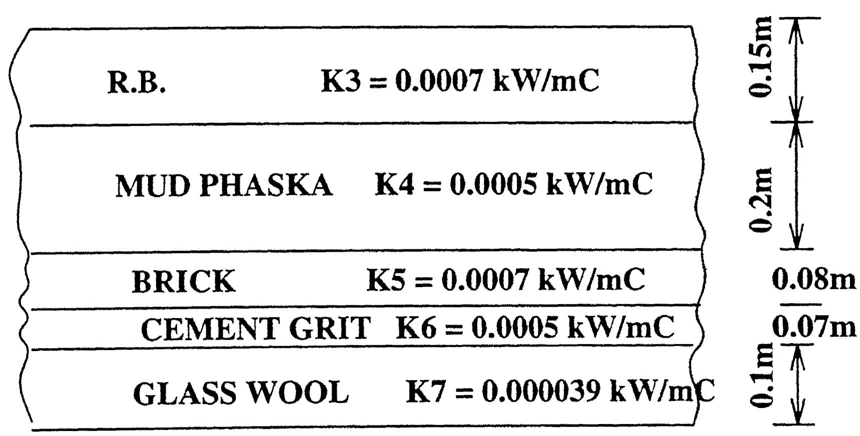

Figure 3.4: Structural detail of ceiling.

Thermal capacity of wall C_{th}, is expressed as·

$$C_{th} = mc = \rho cV = \rho cA\Delta t \tag{3.20}$$

Due to thermal capacity of a structure the following two effects have been observed [17]

1. There is a time lag between the heat transfer at the outside surface and the inside surface.
2. There is decrement in heat transfer due to the absorption of heat by the wall and subsequent transfer of a part of this heat back to the outside air when temperature of ambient is lower.

Considering these two facts, actual heat transfer at any time 't' is given by:

$$\dot{Q}_t = \sum_{j=1}^{5}[U_j A_j (T_{sol} - T_i) + A_j U_j \lambda_j (T_{o(t-\phi 1} - T_{sol}] \tag{3.21}$$

$$\dot{Q}_b = AU(T_b - T_i) \tag{3.22}$$

where. $\dot{Q}_b$ is the basement load and T_b is the underground temperature being taken as 25°C.

Hence.

$$\dot{Q}_s = \dot{Q}_t + \dot{Q}_b \tag{3.23}$$

The value of $\phi 1$ and λ with respect to wall thickness are given in Figure 3.5 and 3.6 respectively.

In the present study equation 3.21 is used to calculate the heat transfer through the structures.

3.3.7 Over-All Heat Transfer Coefficient for Walls and Roof

To get overall heat transfer coefficient U, the value of h_o and h_i are taken from [6], as given below:

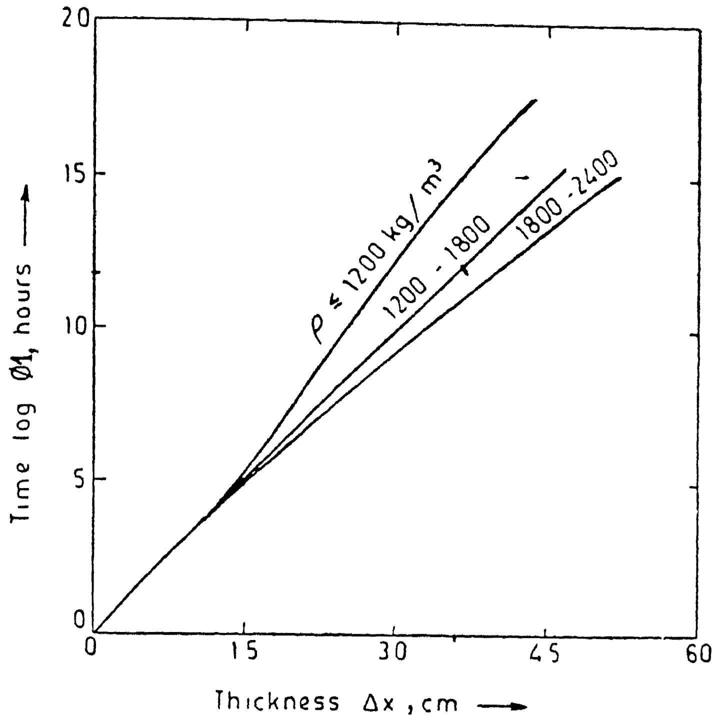

Figure 3.5: Variation of time-lag with wall thickness.

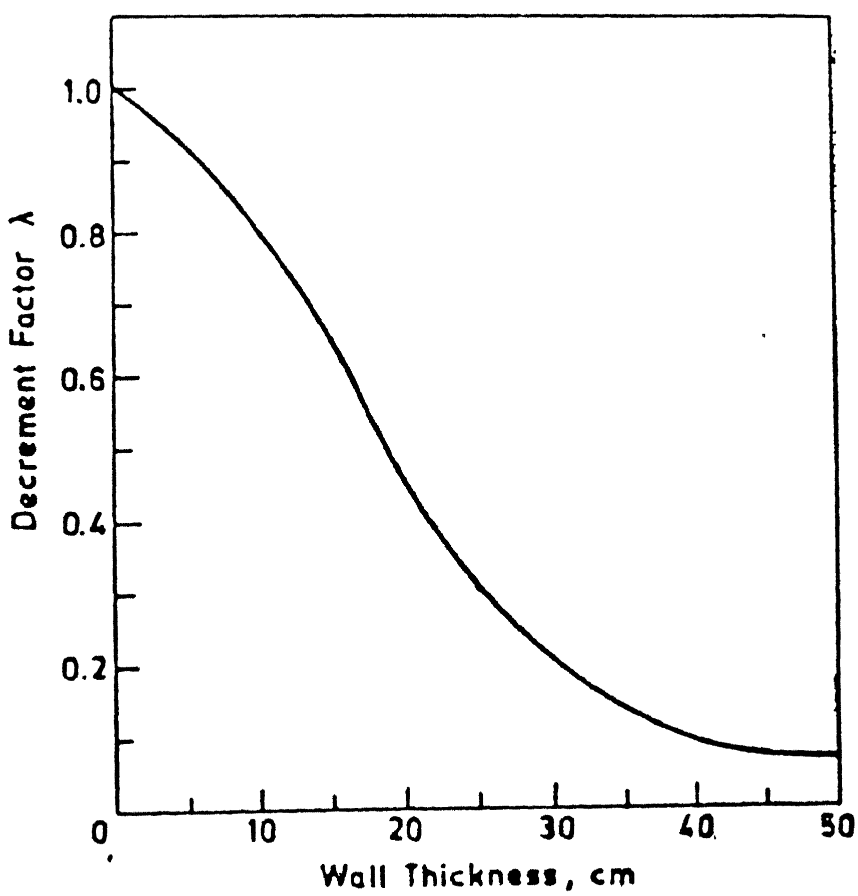

Figure 3.6: Variation of decrement factor with wall thickness.

For exterior wall

$$h_o = 0\ 00391(0.399 + v_w) \tag{3.24}$$

For roof

$$h_o = 0.00141(1.454 + v_w) \tag{3.25}$$

For the inner surfaces of the enclosed space, the convective heat transfer coefficient is found to be the function of the inside air velocity and the temperature difference existing between the room air and inner surface of the wall. In terms of interior air velocity v_i and temperature difference ΔT_i, we get

For interior walls

$$h_i = c_1(\Delta T_i)^{0\ 25} + 0.00391(0.399 + v_i) \tag{3 26}$$

For ceiling

$$h_i = c_2(\Delta T_i)^{0\ 25} + 0.00141(1.454 + v_w) \tag{3.27}$$

where,c_1 and c_2 are constants and its values are

$$c_1 = 1\ 25 \times 10^{-3}, \quad c_2 = 1.45 \times 10^{-3}$$

Now, the value of U for walls and roof taking into account of the time-lag is calculated as follows.

$$q = U[(T_{sol} - T_i) + \lambda T_{o(t-\phi 1)}] = UD \tag{3 28}$$

where,

$$D = (T_{sol} - T_i) + \lambda(T_{o(t-\phi 1)} - T_{em}) \tag{3.29}$$

For the interior wall:

$$q = [c_1(\Delta T_i)^{0\ 25} + 0.00391(0.399 + v_i)]\Delta T_i \tag{3.30}$$

$$q = c_1(\Delta T_i)^{1.25} + h_w \Delta T_i \tag{3.31}$$

where,

$$h_w = 0.00391(0.399 + v_i)$$

Comparing the equation 3.28 and 3.30 we get

$$c_1(\Delta T_i)^{1.25} + h_w \Delta T_i = UD \tag{3.32}$$

But

$$U = \frac{1}{\frac{1}{h_o} + \Sigma\frac{\Delta X_i}{k_i} + \frac{1}{h_i}} \tag{3.33}$$

Putting

$$\Sigma\frac{\Delta X_i}{k_i} + \frac{1}{h_o} = R = \text{ Constant} \tag{3.34}$$

for a given wall and for constant outside wind velocity(v_w) equation 3.32 becomes

$$c_1(\Delta T_i)^{1.25} + h_w \Delta T_i = \frac{D}{\frac{1}{h_i} + R} = \frac{D}{\frac{1}{c_1(\Delta T_i)^{0.25} + h_w} + R} \tag{3.35}$$

$$D = \Delta T_i + Rc_1(\Delta T_i)^{1.25} + Rh_w \Delta T_i \tag{3.36}$$

The Newton-Raphson iterative procedure has been used to get ΔT_i. Accordingly we have

$$F = \Delta T_i + Rc_1(\Delta T_i)^{1.25} + Rh_w \Delta T_i - D \tag{3.37}$$

Thus, after getting value of ΔT_i, h_i is calculated and hence values of U'^s for different walls and roof are calculated

3.4 Infiltration Load

There is always leakage of outdoor air into a building through cracks and openings caused by pressure difference across the boundary surfaces. This leakage is called infiltration load. The exchange of air may cause both types of load namely latent heat load and sensible heat load. The load due to infiltration is given by the following expression:

$$\dot{Q}_{infil} = \frac{V_{room} N_{ach}(H_{o,a} - H_{i,a})}{v_{air}(24 \times 3600)} \tag{3.38}$$

3.5 Lighting and Other Electrical Application

In general the instantaneous rate of heat gain from electric lighting is calculated from

$$\dot{Q}_{light} = Total\ Light\ Wattage \times Cooling\ Load\ Factor\ (CLF) \qquad (3.39)$$

CLF is a function of time of use, type of arrangement, room furnishing etc in present study, a value of 0.88 is taken from[7]

Other electrical appliance used normally is fan and the instantaneous load from the fan is equal to the total rating(Q_{fan})

Therefore,

$$\dot{Q}_{equip} = \dot{Q}_{light} + \dot{Q}_{fan} \qquad (3.40)$$

3.6 Sensible and Respiration Load of Stored Products

In cold storage generally potato is stored. Hence,

$$\dot{Q}_R = \frac{TN \times Resp}{24 \times 3600} \qquad (3.41)$$

During loading period, there is sensible heat load which is equal to

$$\dot{Q}_{sen} = TN \times 1000 \times c_{pp}(T_o - T_i) \qquad (3.42)$$

3.7 Total Load

Total load is sum of all the load, i.e. Solar load, infiltration load, equipment load, respiration load and sensible load.

$$\dot{Q}_{total} = \dot{Q}_{solar} + \dot{Q}_{infil} + \dot{Q}_{equip} + \dot{Q}_R + \dot{Q}_{sen} \qquad (3\ 43)$$

3.8 Solar Load With Surface Evaporation

When surface evaporation is applied to the roof of the building, the temperature of the roof surface approaches to wet-bulb temperature. Hence the solar load on the roof with the surface evaporation is given by

$$\dot{Q}_{roof} = A_R U_R (T_{wb} - T_i) \tag{3.44}$$

Using the procedure outlined in this chapter, the hourly cooling load and hence total yearly cooling load are calculated with and without surface evaporation for "Amar Cold Storage" Chaubepur, Kanpur and the same is compared with the actual load.

Chapter 4

Ammonia-Absorption System a Substitute of Vapour-Compression System for Cold Storage

4.1 Introduction

Cold storage plays significant role in preventing the spoilage of the huge quantity of perishable products by providing an appropriate environment. Thus, it serves the nation in a big way and also helps grower to sell their produce at premium in place of selling the same at throw away prices.

Unfortunately, these days cold storages are facing a serious problem of shortage of power, especially in rural areas. The existing system depends fully upon external power supply on which they have no control. So this calls for development of a system which may operate at disposal of the owner of a cold storage. In the present case ammonia-absorption system is studied to replace ammonia-compression system. Analyses are carried out on the basis of data collected from local cold storages. It is found that operational cost of the ammonia-absorption system is about 60 % to that of existing system. The

detailed result are given in Chapter 5. The compressor of the existing system is replaced by a generator, an absorber, an analyser, a rectifier and a pump as evident from a typical ammonia-absorption system shown in figure 4.1. By this arrangement major components of the compression refrigeration system and their orientation and installation remain unaltered. This aspect convinces cold storage owner for accepting the change. Despite high initial cost compared to the existing system, the additional cost is compensated within five to six years due to saving in the running cost.

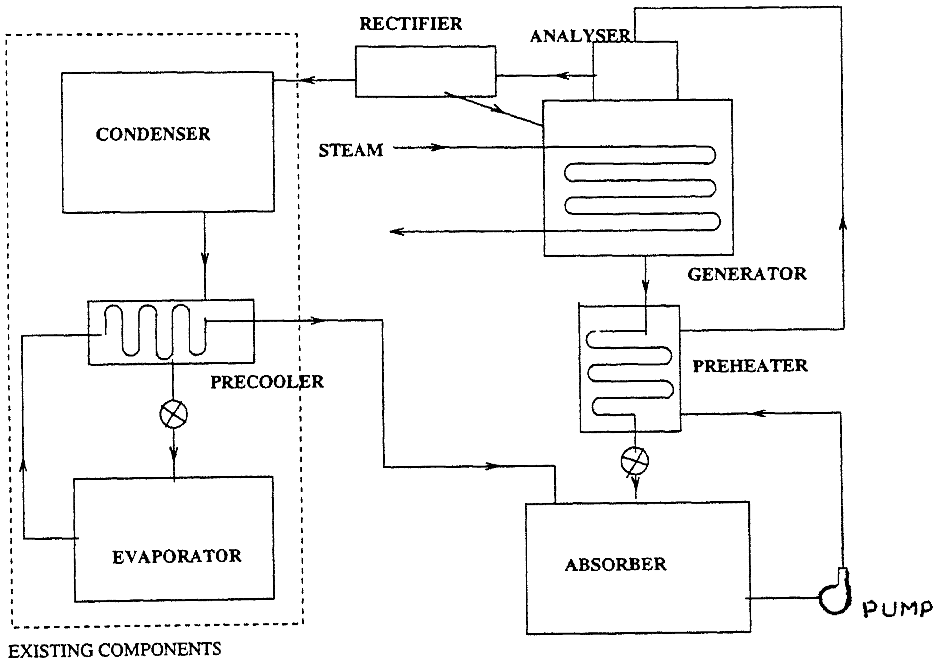

Figure 4.1: A typical ammonia-absorption system for a cold storage.

4.2 Why Cold Storage Should Shift to Vapour-Absorption System ?

As cold storage is necessary for perishable products,the existing erratic power supply leads to spoilage of the commodities. Hence, a self dependent system is the required solution. Moreover, the existing power scenario reveals the dismal position in continuous or uninterrupted power supply to the cold storage. This

leads to procurement of a big Diesel-Generating set at an enormous investment (about 3 to 4 lacs). This additional standby unit raises the cost of preservation load on diesel fuel requirement.

A cold storage generally stores 6000 to 7500 tonnes of potatoes having 100 kW of power consumption. If there are 10 cold storages, the total power requirement becomes 1000 kW, i.e., 1.0 MW. To supply 1.0 MW of power at the cold storage, the installed capacity of the power house should be $1.0 \times 1.25/0.6 = 2.0$ MW as 25 % of power is lost in transmission and plant load factor of power plant is about 60 %. Hence, to get 1.0 MW at the place of use the installed capacity of 2.0 MW requires an investment of Rs. 9 crores. It implies, that the Government has to invest Rs. 90 lacs to meet the energy need of each of cold storages. The net result is the increased cost of potato storage. To reduce the storage cost, the theft of power or tempering of meter reading by some owners cannot be ruled out.

Hence, adoption of ammonia-absorption refrigeration system would not only solve the power problem of cold storage owners, but also help to improve power situation for commercial, industrial and domestic activities Moreover it would be beneficial to both the owner of cold storage and the Government.

4.3 Existing and Modified System

Figure 4.2 exhibits the existing system and the suggested ammonia- absorption system with common components It is evident that only compressor is getting replaced by a generator, an absorber, a pump and a heat exchanger. The rest system remains unaltered. Hence, there is no need of any change inside the cold storage. So the owners may get convinced for the change in their existing system.

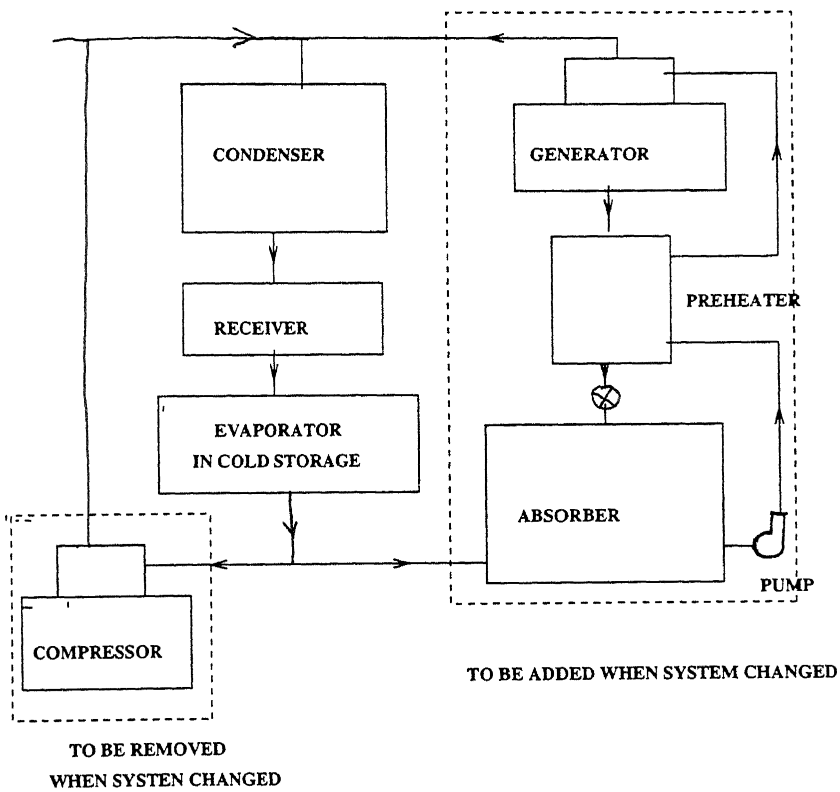

Figure 4.2· A comparision between existing and change needed in the existing system.

Chapter 5

Result and Discussion

This chapter deals with the calculation of cooling loads on the basis of the theoretical and actual temperature variation having effect of heat capacity and time-lag factor of structures For this a generalised computer programme is made based on mathematical formulation discussed in chapter 3. Flow diagram is shown in figure 5.1. To have energy conservation surface evaporation is applied on the roof of cold storage which reduces the roof temperature at one side and nullify the effect of radiation on other side. It is also discussed that decrease in inside temperature of cold storage, there is reduction in cooling load due to decreased respiration load of potatoes. To solve energy problem ammonia vapour-compression is replaced by ammonia-absorption system which reduces operating cost substantially.

5.1 Graphical Representation of Results

5.1.1 Theoretical and Actual Temperature Variation

The actual hourly temperature variation as well as the suggested sinusoidal hourly variation in temperature based on maximum and minimum temperature variation of the day have been shown in figure 3.1. The actual temperature is

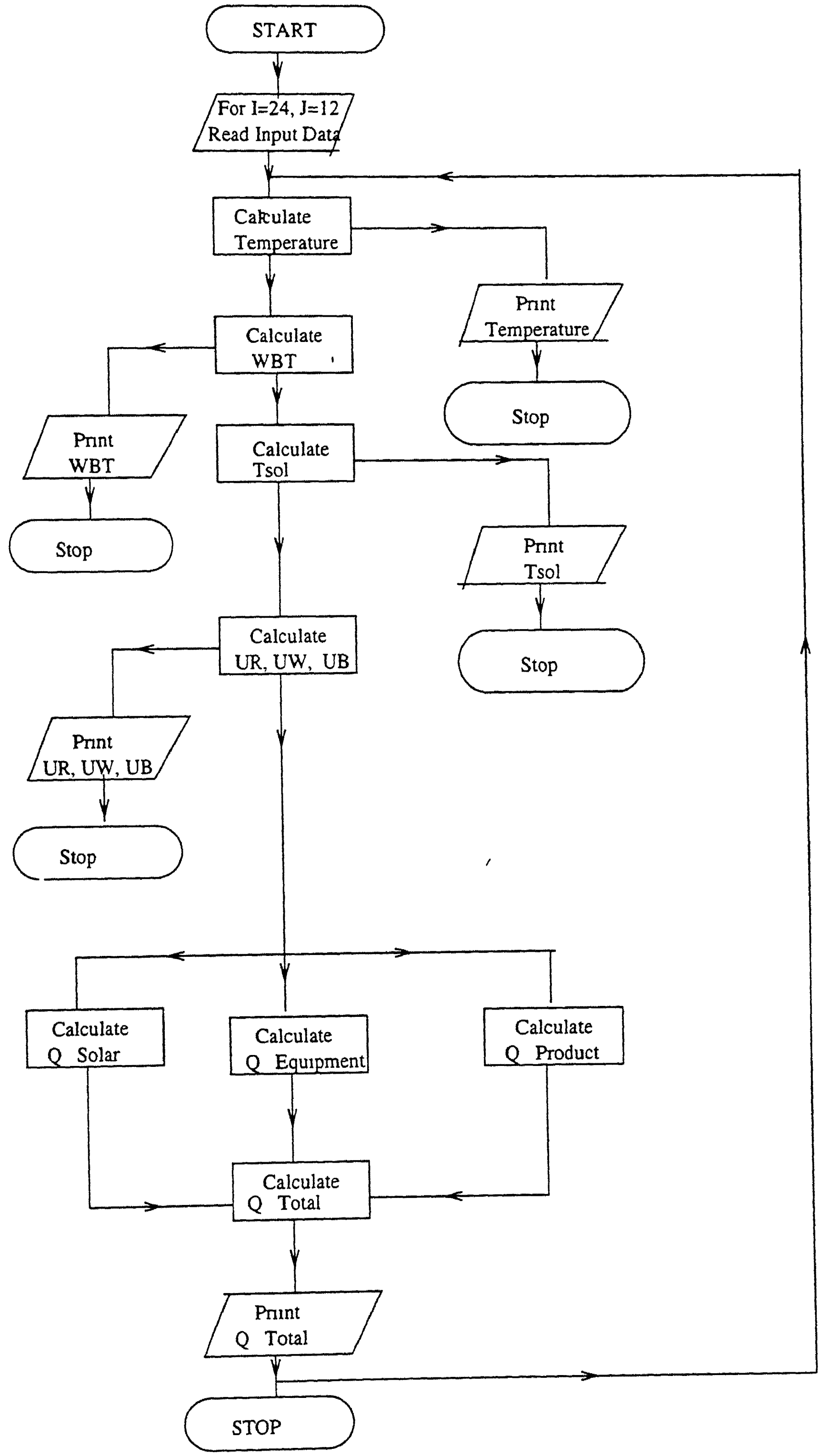

Figure 5.1: Flow chart.

generally seen to be slightly higher than the theoretical value from 8 A.M. to 2:30 P.M., just before peak hour. Thereafter, the actual temperature is lower than that of the theoretical value. The areas under both the curve reveal a small difference (3 to 8 %).

5.1.2 Sol-air Temperature Variation For Different Walls and Roof

The ambient temperature and sol-air temperatures over the different walls and the roof is presented in figure 5.2. The typical result presented is for the month of May. From this plot it is seen that the sol-air temperature for a horizontal surface (roof) is the highest and is maximum at noon, corresponding to the maximum incident solar radiation From the plot it can be inferred that, the heat transfer through the east and west facing walls is more prominent than that of south facing walls. Thus the area of eastern and western walls should be less than that of northern and southern walls.

Figure 5.3 shows the hourly variation in wet-bulb and temperature of different walls and roof for a particular day. The wet-bulb temperature is much lower than that of temperatures of different walls and remains almost constant. This vindicates that the evaporation in nature is close to an adiabatic saturation process. Figure 5.4 shows temperature variation of roof and wet-bulb temperature. If surface evaporation is applied on the roof, the roof surface temperature tends to wet-bulb temperature. Thus, we see that there is substantial reduction in roof surface temperature and hence reduced cooling load by applying surface evaporation Figure 5.5 exhibits variation in dry-bulb temperature and wet-bulb temperature for a typical month of May.

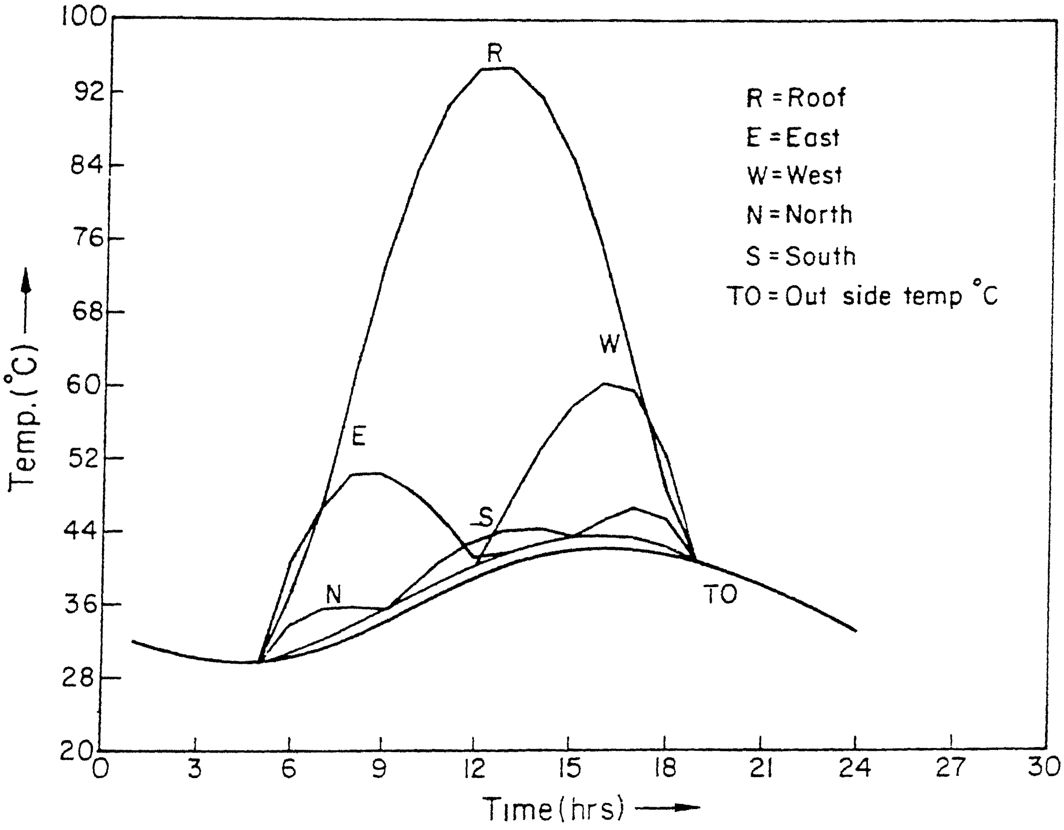

Figure 5.2: Variation of ambient temperature and sol-air temperature for walls and roof for the month of may.

5.1.3 Variation of Wet-Bulb Temperature and Relative Humidity

Figure 5.6 shows variation of relative humidity with time for the month of May. The relative humidity is minimum at about 3 P.M. when dry-bulb temperature

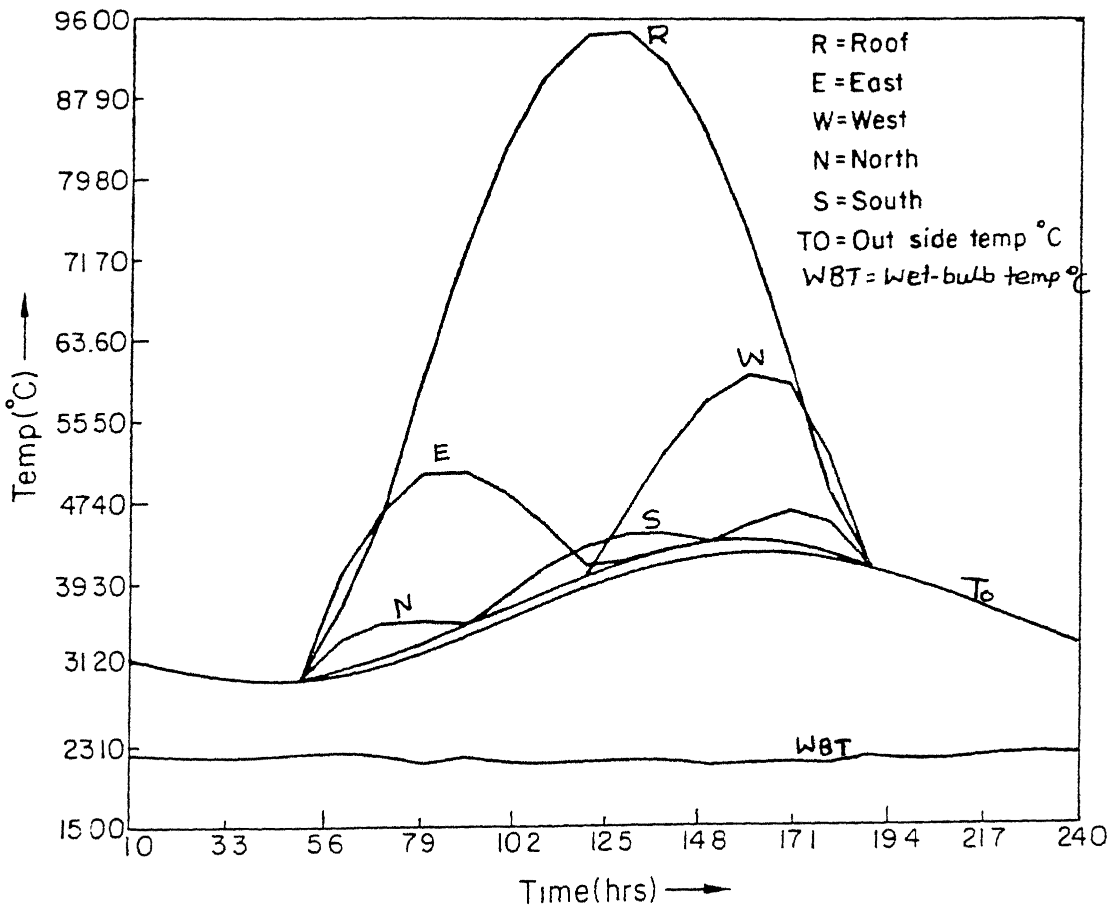

Figure 5.3: Variation of ambient temperature and sol-air temperature for walls and roof along with wet-bulb temperature for the month of may

is maximum.The nature of the curve is the same for each month. This helps to keep the wet-bulb temperature almost constant (variation range 3 tom 5 °C

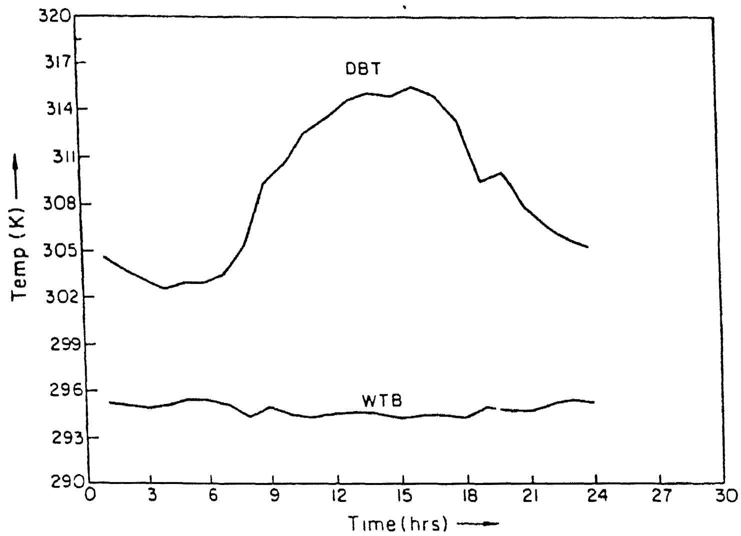

Figure 5.4: Variation of temperature of roof along with wet-bulb temperature.

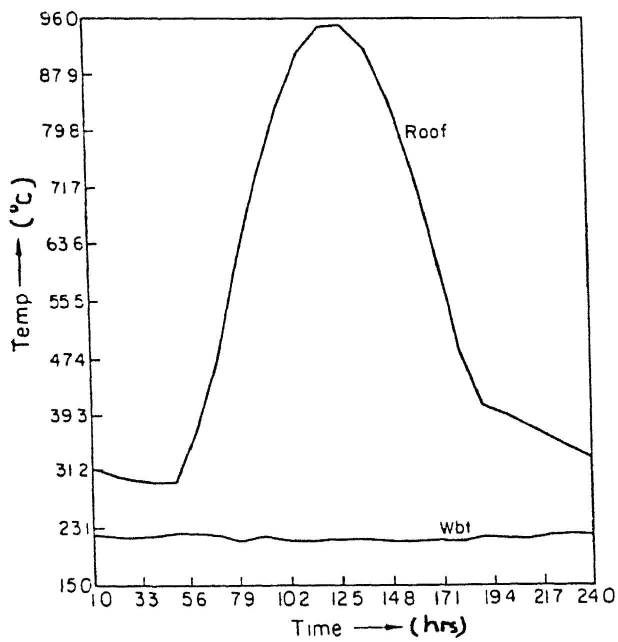

Figure 5.5: Variation in sol-air and wet-bulb temperature for the month of may.

) as shown in figure 5 7. The wet-bulb temperature variation is maximum for the month of November, December and January as during that time maximum minimum temperature and relative humidity variation is maximum. Where as wet-bulb temperature is almost constant for the months of July and August when maximum minimum temperature variation is minimum.

5.1.4 Solar Load With Surface Evaporation

Figure 5.8 shows hourly variation of solar load with and without surface evaporation where as figure 5.9 shows monthly variation of solar load with and without surface evaporation. From the figure it is clear that hourly variation of solar load is almost constant with surface evaporation. This is supported from figure 5.7 where the hourly variation in wet-bulb temperature is found to be very small as compared to the ambient dry-bulb temperature.However, when there is no surface evaporation, it is maximum at around 8 A.M. due time lag factor of about 19 hours. Solar load is maximum for the month of June in both with and without surface evaporation.

5.2 Tabulated Results

Table 5.1 shows average solar load per day with and without surface evaporation, respiration load per day for each month and total load with and without surface evaporation per month. Generally, in cold storage, loading starts after 15^{th} February and it is filled to half of its capacity during that month. So, in February total load is calculated for 14 days and for half of the capacity. Generally, cold storage is full by the end of March but in some cases it may go also in first week of April. In these calculations it is assumed that, cold storage is full by the end of March. During February and March sensible load of potato is also included in respiration load. Inside temperature is maintained at 2 °C. After March there is no sensible heat load. Hence respiration load is constants.

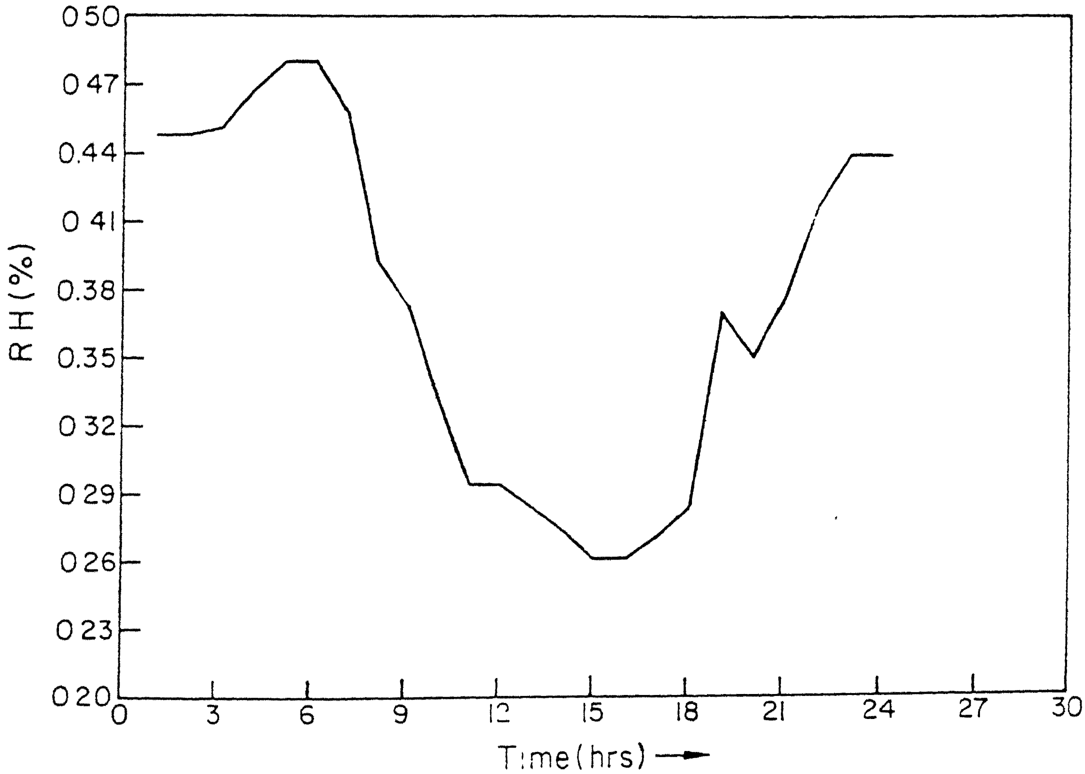

Figure 5.6: Hourly variation in relative humidity for may.

Table 5.2 shows total cooling load with and without surface evaporation, percentage saving in cooling load and saving in terms of rupees. The cost of equipment for surface evaporation is about Rs. 75,000 which will be recovered

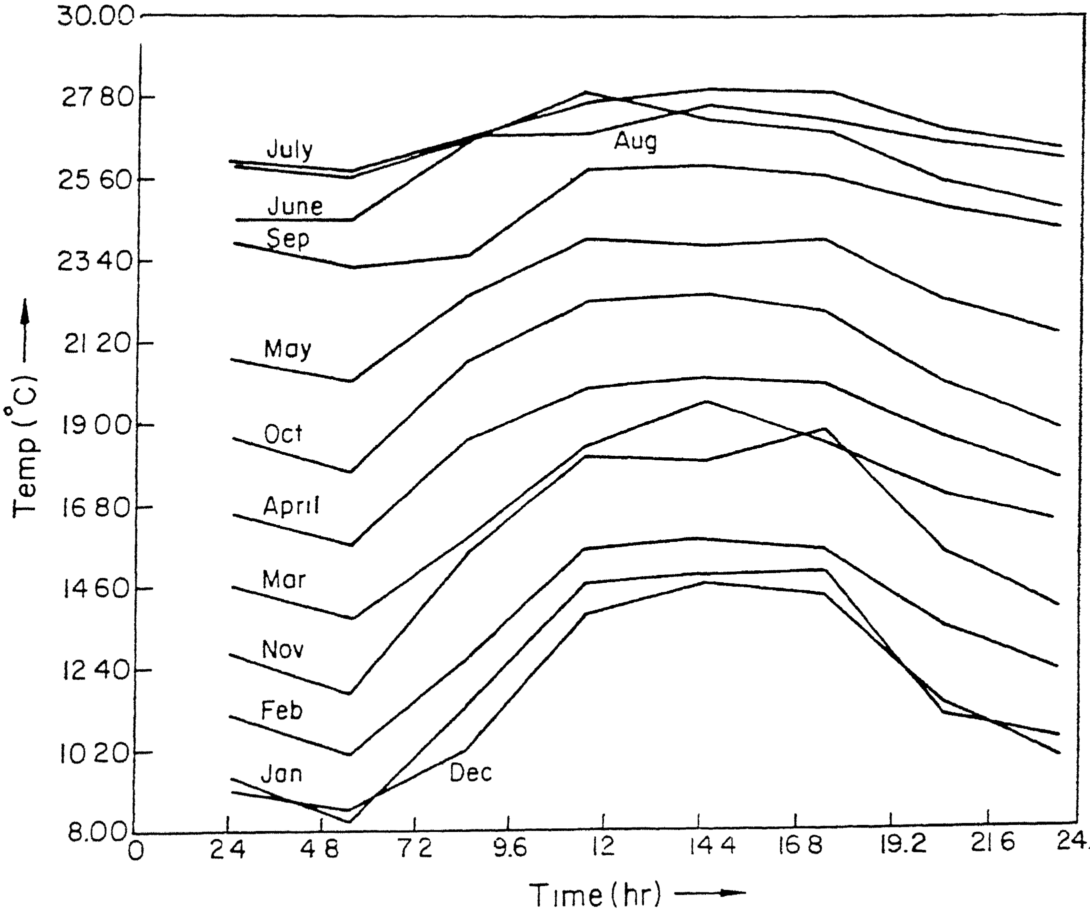

Figure 5.7: Hourly variation of wet-bulb temperature for different months.

within one year.

Table 5.3 shows the reduction in cooling load as storage temperature decreases. This is for the month of May. This is due to reduced heat of respiration. Heat

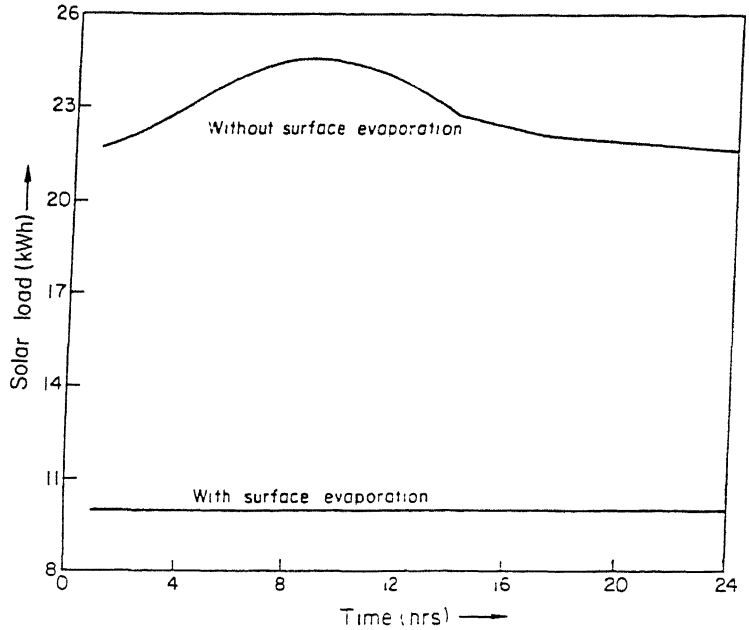

Figure 5.8· Hourly variation of solar load in may.

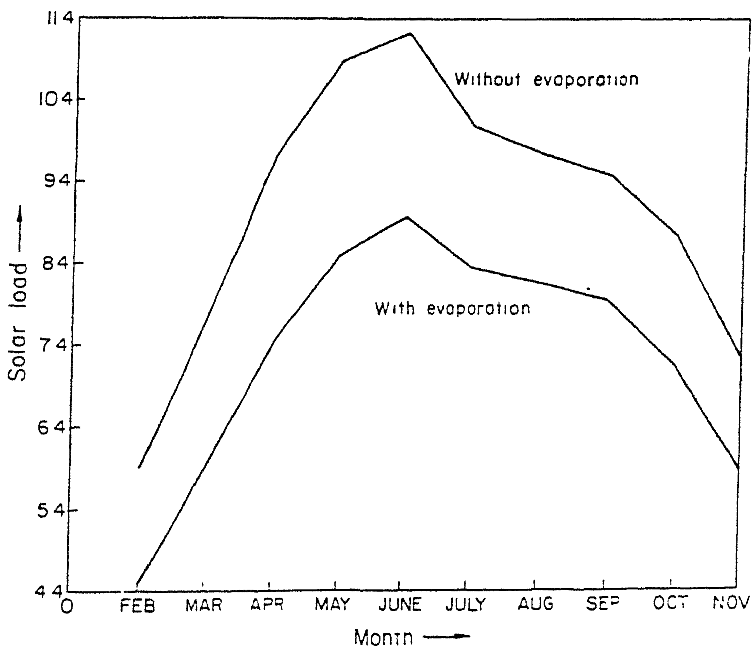

Figure 5.9: Monthly variation of solar load.

of respiration is taken from [20]. The same is given at temperature 4.5 °C and 0 °C. Linear interpolation is used for heat of respiration at other temperatures.

Table 5.4 shows actual cooling load supplied by Amar Cold Storage, Chaubepur, Kanpur and cooling load calculated using computer programme. The data supplied by cold storage is meter reading, i.e., energy consumed by compressor. The same is compared with calculated load by converting in terms of cooling load by multiplying with COP.

Actual cooling load of February is much less than the calculated load. This is due to fact that in calculated load it is assumed that half of loading is completed in February but the same is not actually true. In calculation it is assumed that loading is over by the end of March. But actually loading has gone to the month of April. That is the reason for more actual cooling load than the calculated load in the month of April. For the other month cooling load is comparable

Table 5.1: Solar, respiration and total load.

Month	Solar Load		Respiration Load	Total Load	
	Without Evap.	With Evap.		Without Evap.	With Evap.
	kWh/day	kWh/day	kWh/day	kWh/month	kWh/month
Feb.	592.22	449.59	2222.22	42216.56 for 15 days	40077.71 for 15 days
March	774.84	591.69	4444.44	156578.63	151084.1
April	972.42	744.8	3413.33	131572.64	124744.02
May	1091.21	851.51	3413.33	135136.33	127945.34
June	1126.29	898.65	3413.33	136188.81	129359.55
July	1011.9	836.98	3413.33	132576.92	127509.51
Aug.	977.14	817.2	3413.33	131714.17	126911.11
Sept.	949.99	795.82	3413.13	130899.64	126274.48
Oct.	876.42	716.54	2666.67	106292.48	101496.24
Nov.	724.68	587.82	1706.67	72940.27	68834.48

Table 5.2: Cooling load with and without surface evaporation and saving.

Total cooling load without surface evaporation	1176296.4 kWh
Total cooling load with surface evaporation	1124241.5 kWh
Saving in cooling load	52054.9 kWh
Percentage of saving	4.42 %
Saving in terms of Rs.	Rs. 78082.32

Table 5.3: Reduction in cooling load due to reduced storage temperature.

Storage Temperature	Heat of Respiration	Total Load	
		Without Evap.	With Evap.
^oC	kJ/day-ton	kWh/month	kWh/month
4 5	2720	179334.36	168167.00
4.0	2556.7	167027.7	159859.89
3 0	2230	150409.1	143240.41
2 0	1903	135136.33	127945.3
1.0	1579 7	117177.52	110006.98
0.0	1250	100559 09	93389.69

Table 5.4: Actual and calculated cooling load.

Month	Calculated Load(kWh/month)	Actual Load(kWh/month)		
		cop 3.0	cop 3.25	cop 3.5
Feb.	42216.56	16920	18330	19740
March	156578.63	131577	142541.25	153506.5
April	131572.64	164325	178018.75	191712.5
May	135136.33	131316	141259	153202
June	136188.81	121101	131192.75	141284.5
July	132576.92	119151	129680.25	139009.5
Aug.	131714.17	116835	126571.25	136307.5
Sept	130899 64	122301	132492.75	142684.5
Oct.	106292.48	97389	105504.75	113620.5
Nov.	72940.27	65055	70476.25	75897.5
	1176276.39	1085970	1176467.8	1266965

Table 5.5 shows the saving in operational cost due to adoption of ammonia-absorption system in place of vapour-compression system. This calculation is made from the energy data supplied by local cold storages which is given in Appendix D. COP of the existing vapour-compression system is taken as 3.0 where as COP of proposed ammonia -absorption system is taken as 0.6 based on primary energy. Primary energy used for calculation is coal. Calorific value of coal is taken as 16,000 kJ/kg, and its cost is Rs. 1500 per tonne. The cost of power is taken as Rs 3.75/kWh as supplied by cold storage owner.

Table 5.5: Saving in operational cost in ammonia-absorption system against vapour-compression system.

Kanaoj Cold Storage, Kanauj	
Storing Capacity	7000 tonnes
Electric Bill	Rs. 1447500/year
Cost of Coal	Rs. 900000/year
Saving	Rs. 547000/year
Percentage Saving	37.8 %
Amar Cold Storage Chaubepur, Kanpur	
Storing Capacity	6400 tonnes
Electric Bill	Rs. 1357470/year
Cost of Coal	Rs. 825000/year
Saving	Rs. 532470/year
Percentage Saving	39.25 %
Raj Cold Storage, Kanpur	
Storing Capacity	11000 tonnes *
Electric Bill	Rs. 3053780/year
Cost of Coal	Rs 1875000/year
Saving	Rs. 1178780/year
Percentage Saving	38.6 %

* This cold storage has an ice plant also.

Chapter 6

Conclusions and Suggestions

6.1 Conclusions

1 The total cooling load has been calculated on the basis of hourly temperature variation based on T_{max} and T_{min} for a given day. The same compared with actual cooling load based on actual hourly temperature variation.

2 The overall heat transfer coefficient has been calculated accurately by considering the dependence of inside convective heat transfer on the inside surface temperature and air motion. Newton-Raphson method has been used to calculate the same.

3. Surface evaporation is applied to the roof of the cold storage. It renders reduced cooling load to the tune of 4.5 %. This percentage is found to be low due to dominance of the respiration load of potatoes.

4 By decreasing the storage temperature, the cooling gets reduced due to reduced heat of respiration of potatoes at lower temperature.

5. To solve the power problem in cold storage, the replacement of the existing ammonia-compression system by ammonia-water vapour-absorption system is recommended. This would enable the use of primary energy for refrigeration.

6. By using ammonia-absorption system, the operating cost is reduced by 40 %.

7. The recommended system will operate at the disposal of the owner of cold storage. Hence, better quality of the stored products.

6.2 Scope for Future Work

1. The surface evaporation should be applied to the roof and actual reduction in cooling load should be measured.

2. Economic analysis can be done including the cost of surface evaporation and absorption system.

3. Ammonia-absorption system is to be designed and the same is installed in cold storage in order to demonstrate its salient features.

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Appendix A

Temperature and Relative Humidity of Outside Air

Table A.1: Average temperature and relative humidity of outside air at Kanpur, January to June.

Time	January		February		March		April		May		June	
	T	ϕ	T	ϕ	T	ϕ	T	ϕ	T	ϕ	T	ϕ
hrs	$^\circ C$	%	$^\circ C$	%	$^\circ C$	%	$^\circ C$	%	$^\circ C$	%	$^\circ C$	%
2 30	10 28	86 2	12 9	81 45	18 65	65 5	22 9	52 8	28 05	52 25	29 25	68 5
5 30	9 5	88 9	11 4	85 3	16 65	73 0	20 8	59 4	26 05	58 8	28 3	74 0
8 30	11 35	87 0	14 7	79 75	21 1	59 0	27 8	41 4	32 8	41 4	32 65	63 1
11 30	18.75	58 5	23.0	46 0	29 1	35 3	35 15	23 8	39 2	28 4	37 65	48 5
14 30	22 05	45.15	25 7	35 3	33.65	26 3	37 25	20 0	41 5	22 7	39.25	40 25
17 30	19 9	55	23 85	41 9	30 65	30 35	35 3	23 75	39 6	27 25	37.6	44 3
20 30	13 65	77.1	17 15	67 3	23 95	50 45	28 55	38.55	32 6	41 3	32 8	56 5
23 30	11 5	83 15	14 7	76 5	20 95	63 0	25 35	45.95	29 78	48 0	30 4	64 0

Table A.2: Average temperature and relative humidity of outside air at Kanpur, July to December.

Time	July		August		September		October		November		December	
	T	ϕ	T	ϕ	T	ϕ	T	ϕ	T	ϕ	T	ϕ
hrs	^{o} C	%	^{o} C	%	^{o} C	%	^{o} C	%	^{o} C	%	^{o} C	%
2 30	27 5	90 0	26 85	93 55	25 25	89 45	20 3	85 7	14 3	84 65	10 7	85 85
5 30	26 95	91 95	26 55	93 5	24 3	91 8	18 95	88 9	12 7	89 3	9 2	88 7
8 30	29 0	84 55	28 2	89 55	27 51	72 35	24 3	72 9	17 75	79 85	13 1	82 0
11 30	32 0	72 4	30 8	80 15	31 25	65 75	31 8	44 15	27 2	41 35	22.4	43 1
14 30	33 2	68 25	31 2	76 3	32 5	60 3	33 2	40 05	29 3	32 65	24 65	34 35
17 30	32 45	71 85	30 0	81 25	30 5	68 68	28 55	57 15	23 3	59 15	20 3	57 3
20 30	29 6	81 85	28 15	89	27 25	82 6	23 5	74 05	18 2	76 4	13 3	77 0
23 30	28 4	86 1	27 35	91 5	26 05	86 7	21 05	81 75	15 85	82 5	12 05	82 65

Table A 3: Maximum and minimum temperature of outside air at Kanpur.

Month	Maximum Temperature	Minimum Temperature
	^{o} C	^{o} C
January	23.1	8.0
February	26.5	10.05
March	32.65	15.4
April	38.15	19.9
May	41.9	25.18
June	41.35	27.6
July	34.55	26.25
August	32.62	26.04
September	33.6	23.85
October	34.36	18.23
November	30.15	11.75
December	26.05	8.1

Appendix B

Values for Solar Intensity Calculation

Month	Equation of Time minute	Declination Angle degree	A W/m^2	B	C
Jan	-11.2	-20	1230	0.142	0.058
Feb	-13.9	-10.8	1214	0.144	0.600
March	-7.5	0.0	1185	0.156	0.071
April	1.1	11.6	1135	0.180	0.097
May	3.3	20.0	1103	0.196	0.121
June	-1.4	23.45	1088	0.205	0.134
July	-6.2	20.6	1085	0.207	0.136
August	-2.4	12.3	1107	0.201	0.122
Sept.	7.5	0 0	1151	0.177	0.092
Oct.	15.4	-10.6	1192	0.160	0.073
Nov.	13.8	-19.8	1220	0.149	0.063
Dec.	1.6	-23.45	1233	0.142	0.057

A =Apparent solar irradiation at air mass = 0, W/m^2

B= Atmospheric extinction coefficient

C= Diffuse radiation factor

Appendix C

Cold Storage Particulars for Cooling Load Calculation

Cool Room Size	$37.8\times 31.0 \times 16.8\ m^3$.
Area of East Facing Wall	$31.0 \times 16.8\ m^2$.
Area of West Facing Wall	$31.0 \times 16.8\ m^2$.
Area of North Facing Wall	$37.8 \times 16.8\ m^2$.
Area of South Facing Wall	$31.0 \times 16.8\ m^2$.
Area of Roof	$37.8 \times 31.0\ m^2$.
Area of Base	$37.8 \times 31.0\ m^2$.
Thickness of Wall	0.56 m.
Thickness of Roof	0.60 m.
Capacity of Cold Storage	6400 tonnes.
Number of Fans	84.
Rating of Fans	40 watts.

Appendix D

Energy Data of Cold storages

Table D.1. Power taken by compressors of Kanauj Cold Storage, Kanauj of capacity 7000 tonnes

Month	Compressors Power kWh
Feb.	22000
March	47000
April	47000
May	52000
June	57000
July	47000
August	52000
Sept.	32000
Oct.	17000
Nov.	13000

Table D.2: Power taken by compressors of Amar Cold Storage, Chaubepur, Kanpur of capacity 6400 tonnes

Month	Compressors Power
	kWh
Feb.	5640
March	43859
April	54775
May	43772
June	49367
July	39717
August	38948
Sept.	40767
Oct.	32463
Nov.	21685

Table D.3: Power taken by compressors of Raj Cold Storage, Kanpur of capacity 11000 tonnes

Month	Compressors Power
	kWh
Jan.	15325
Feb.	15690
March	81943
April	134437
May	127266
June	116033
July	75467
August	86637
Sept.	80850
Oct.	45810
Nov.	21453
Dec.	13430

Appendix E

Storage Condition and Properties of Potato

Storage Temperature	2 ^{o} C.
Inside Relative Humidity	85 %.
Freezing Point	-1.7 ^{o} C.
Sp. Heat Above Freezing Point	3.6 kJ/kg ^{o} C.
Sp. Heat Below Freezing Point	1.97 kJ/kg ^{o} C.
Heat of Respiration	1250 kJ/day-ton (at 0 ^{o} C)
	2720 kJ/day-ton (at 4.5 ^{o} C)

Appendix F

Properties of Moist Air

$$p_s = \frac{221\ 287}{exp[\{7\ 21379+(1\ 152\times10^{-5}-4\ 787\times10^{-9}T_{db})(T_{db}-483\ 15)^2\}\{647\ 31/T_{db}-1\}]}$$

$$p_v = p_{wb} - \frac{(p_a - P_{wb})(T_{db} - T_{wb})}{1940 - 1\ 44T_{wb}}$$

$$RelativeHumidity(\phi) = \frac{p_v}{p_s}$$